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Control of Ground Coupled Heat Pump Systems in Offices to Optimally Exploit Ground Thermal Energy Storage on the Long Term

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1. Introduction

In Belgium (and most European countries) office buildings require both heating and cooling to maintain the requested thermal comfort throughout the entire year. Eco-innovative installation technologies and control strategies are expected to satisfy the heating and cooling demand in a highly efficient way. The combination of thermally activated building systems (TABS), allowing both low temperature heating and high temperature cooling, and ground coupled heat pumps (GCHP) represents one of these eco-innovative technologies.

The GCHP system extracts and injects heat from and to the ground by means of vertical borehole heat exchangers (BHEs). This way, the ground can be exploited for seasonal thermal energy storage. However, the building heating and cooling loads are usually not in balance. The question arises how to operate the GCHP system such that a long term sustainable ground storage operation is guaranteed. Recent research on optimal control of GCHP focuses either on the short term objectives (i.e. thermal comfort and short term energy cost, making abstraction of the ground storage dynamics) [1], or on the long term objectives (i.e. ground thermal balance and annual energy cost, making abstraction of the building dynamics) [2].

This paper presents an innovative strategy which combines the short and the long term objectives. An optimization approach is presented, which integrates the thermal dynamics of both the building and the ground storage volume to answer three questions: (1) What are the characteristics of the optimal heating and cooling duties distribution over the heating/cooling devices in order to achieve yearly cyclic ground temperatures behavior while satisfying thermal comfort requirements; (2) What are the differences between two installation concepts for Passive Cooling- (ground coupled, primary cooling) and Active Cooling- (air coupled, back-up cooling) devices when they run simultaneously in serial connection; (3) How does the downsizing of the Active Cooling installation influence the system operation and the primary energy use.

2. Materials and method

The system investigated is an office building equipped with TABS, which is fed by primary and back-up heating and cooling installations. The primary heating installation is an On/Off GCHP connected to a borefield, which consists of vertical BHEs. The action of this installation will be further referred to as Heat Pump heating (HP). A gas boiler is considered as back-up heating installation, which will be called Auxiliary heating (AUX). The primary cooling installation consists of a heat exchanger for passive cooling, which injects heat in the ground through the borefield. Its action will be further referred to as Passive Cooling (PC). As back-up cooling installation an On/Off water to water chiller connected to a cooling tower is assumed, which will be called Active Cooling (AC). The system scheme is depicted in Figure 1.

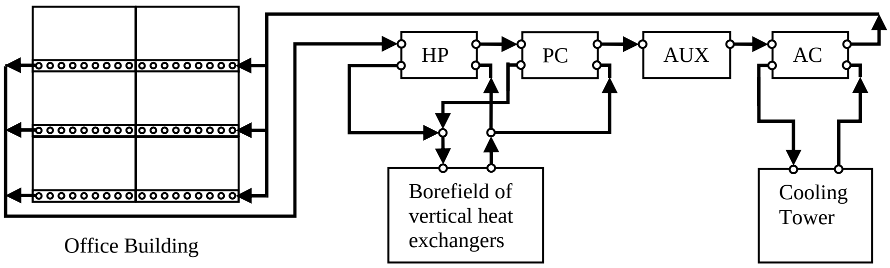

Figure 1. Schematic presentation of the system investigated

The office zones and the borefield are represented by their discrete state space models in Matlab [3] with a discrete time step of 1 hour. An office zone model of 4[th] order is used [4], for which the states are: the concrete core temperature of the TABS – T_{CC}, the office zone air temperature – T_Z, the internal walls temperature – T_{WI}, and the external walls temperature – T_{WO}. Office model inputs are: the supply water temperature – T_{WS}, the ventilation air temperature – T_{VS}, the ambient air temperature – T_{AMB}, the internal gains – $\dot{Q}_{INT}$, and the solar gains – $\dot{Q}_{SOL}$. The borefield is considered as a collection of vertical BHEs that do not thermally interact with each other. Therefore, the borefield model consists of several uncoupled vertical BHE models. A BHE model of 3[rd] order is used [5], for which the states are the mean ground temperatures T_j *, for* $j=1,2,3$, of three concentric cylinders of ground around the borehole. Model inputs are: the extraction heat flow rate – $\dot{Q}_{BF}$ and the undisturbed ground temperature $T_{GR} = 10°C$ located at the outer boundary of the storage volume. Model output is the outlet temperature of the circulating brine – T_{BRINE}. The building model and the BHE model are represented by the state space equations (1), (2), and (3), where i is the corresponding time step.

$$\begin{bmatrix} T_{CC}(i+1) \\ T_Z(i+1) \\ T_{WI}(i+1) \\ T_{WO}(i+1) \end{bmatrix} = A_{BD} \begin{bmatrix} T_{CC}(i) \\ T_Z(i) \\ T_{WI}(i) \\ T_{WO}(i) \end{bmatrix} + B_{BD} \begin{bmatrix} T_{WS}(i) \\ T_{VS}(i) \\ T_{AMB}(i) \\ \dot{Q}_{INT}(i) \\ \dot{Q}_{SOL}(i) \end{bmatrix} \tag{1}$$

$$\begin{bmatrix} T_1(i+1) \\ T_2(i+1) \\ T_3(i+1) \end{bmatrix} = A_{BHE} \begin{bmatrix} T_1(i) \\ T_2(i) \\ T_3(i) \end{bmatrix} + B_{BHE} \begin{bmatrix} \dot{Q}_{BF}(i) \\ T_{GR} \end{bmatrix} \tag{2}$$

$$T_{BRINE} = C_{BHE} \begin{bmatrix} T_1(i) \\ T_2(i) \\ T_3(i) \end{bmatrix} + D_{BHE} \begin{bmatrix} \dot{Q}_{BF}(i) \\ T_{GR} \end{bmatrix} \tag{3}$$

The primary and the back-up heating and cooling devices are represented by their efficiencies and the heat flows that they provide. The heat pump is modeled by a constant approximation of its coefficient of performance – COP_{HP}, which links the heat flow rate at the condenser side – $\dot{Q}_{HP}$ to the heat flow rate at the evaporator side – $\dot{Q}'_{HP}$ as shown in Eq. (4), where i is the corresponding time step.

$$\dot{Q}_{HP}(i) = \frac{COP_{HP}}{COP_{HP}-1} \dot{Q}'_{HP}(i) \tag{4}$$

The chiller for active cooling is modeled analogically but only the heat flow rate at the evaporator side is incorporated – $\dot{Q}_{AC}$. The chiller efficiency – COP_{AC} is defined as the ratio between the rejected heat

($\dot{Q}_{AC}$) and the consumed electrical energy. Passive cooling is characterized by the heat flow rate injected in the ground – $\dot{Q}_{PC}$. The passive cooling efficiency (η_{PC}) is defined as the ratio between $\dot{Q}_{PC}$ and the electricity needed to drive the circulation pumps to feed the passive cooling heat exchanger. The Auxiliary heating is characterized by the delivered heat flow rate – $\dot{Q}_{AUX}$. The gas boiler efficiency (η_{AUX}) is defined as the ratio between $\dot{Q}_{AUX}$ and the primary energy corresponding to the consumed gas.

All models of system components are integrated into a global system model by Eqs. (5) and (6):

$$\sum \dot{Q}_{TABS}(i) = \dot{Q}_{HP}(i) + \dot{Q}_{AUX}(i) - \dot{Q}_{PC}(i) - \dot{Q}_{AC}(i) = UA_{T_{WS}T_{CC}} * (T_{WS}(i) - T_{CC}(i)) \tag{5}$$

$$\sum \dot{Q}_{BHE}(i) = \dot{Q}'_{HP}(i) - \dot{Q}_{PC}(i) = \frac{COP_{HP} - 1}{COP_{HP}} \dot{Q}_{HP}(i) - \dot{Q}_{PC}(i) = \dot{Q}_{BF}(i) \tag{6}$$

where $\sum \dot{Q}_{TABS}(i)$ is the total heat flow transferred to the TABS, in the corresponding time step i, and $\sum \dot{Q}_{BHE}(i)$ is the total heat flow extracted from the ground by the ground coupled devices. The translation of $\sum \dot{Q}_{TABS}(i)$ to $T_{WS}(i)$, in order to connect all devices to the building model, is carried out by the UA-value of the heat transfer between the supply water and the concrete core. The dynamics of the devices and the dynamics of the heat and cold distribution system are neglected since their time constants are smaller than the sampling time of 1 hour. Hence, the heat flow from all devices to the supply water is assumed equal to the heat flow from the supply water to the concrete core as expressed in Eq. (5).

Sizing of the system components is based on dynamic calculation of the office building heating and cooling loads for a reference year. The conditioned office area is 3600 m^2, corresponding to 300 office zones of 12 m^2 each. The office zone model considers 2 such zones, so their loads are scaled by 150 to represent the whole office building. The GLHEPro tool [6] is used to define the required number of BHEs, based on the annual heating and cooling loads of the building. In a first design scenario, the borefield is sized to entirely cover the heating and cooling loads (respectively through HP operation and PC). The result from the sizing tool for this scenario is 33 single BHEs of 120 m each. The capacity limitations of the primary heating and cooling devices equal the peak heating and cooling loads of the building. In a second design scenario, the borefield is downsized to 17 BHEs to investigate the operation of the back-up heating and cooling devices, respectively auxiliary heater and active cooling. Their capacity limitations are also subject of investigation. Initially they are set to the capacities of the corresponding primary devices.

The operation of the modeled GCHP system is defined by means of an optimization procedure minimizing an objective function, which is subject to constraints. The method is called “Short and Long Term Optimization” (SALTO) since it combines optimization at short term (system’s daily dynamics, using a sampling time of 1 hour) and long term (ground dynamics over 1 or more years, using 8760 or more time samples). This implies that the decision variables, which minimize the objective function, are contained in 16 vectors, each of 8760 time samples, representing the controllable inputs, the outputs, and the states of the subsystem models, as well as dummy variables representing thermal discomfort. Fixed data for a reference year are used for the non-controllable inputs of the building- and ground models. Meteorological data from [7] are used to obtain one year profiles for ambient air temperature T_{AMB} and solar radiation gains $\dot{Q}_{SOL}$. To represent the internal gains $\dot{Q}_{INT}$ scheduled profiles are used for heat gains caused by office occupancy and appliances. The ventilation supply temperature T_{VS} coincides with the lower comfort boundary T_{MIN} of the zone temperature. The undisturbed ground temperature T_{GR} is assumed yearly constant at 10°C. The optimization problem is formulated numerically in Matlab with the YALMIP interface [8] where the IBM ILOG cplex solver [9] is called to compute the optimal solution for the decision variables.

The objective function J (Eq. (7)) comprises linear terms of operation costs of the heating and cooling devices along the 1 year horizon of system operation, with Δt_i = 1 hour.

$$J = \sum_{i=1}^{8760} \left[C_{EL}(i) \frac{\dot{Q}_{HP}(i)}{COP_{HP}} + C_{GAS} \frac{\dot{Q}_{AUX}(i)}{\eta_{AUX}} + C_{EL}(i) \frac{\dot{Q}_{PC}(i)}{\eta_{PC}} + C_{EL}(i) \frac{\dot{Q}_{AC}(i)}{COP_{AC}} \right] \Delta t_i \tag{7}$$

In Eq. (7) $C_{EL}(i)$ denotes the hourly varying electricity price and $C_{GAS}(i)$ – the gas price. The zone air temperature T_Z is constraint (Eq. (8)) between lower and upper thermal comfort boundaries, $T_{MIN}(i)$ and $T_{MAX}(i)$, which depend on the ambient air temperature T_{AMB} according to the EN15251 standard [10] as for office room Category II.

$$T_{MIN}(i) - \varepsilon_1(i) \le T_Z(i) \le T_{MAX}(i) + \varepsilon_2(i) \tag{8}$$

In Eq. (8) dummy decision variables $\varepsilon_1(i)$ and $\varepsilon_2(i)$ are introduced to represent deviations from the comfort range. According to the standard [10], a total deviation of 0.4 Kelvin hours (Degree hours) is allowed during the daily occupation period. This constraint is implemented by Eq. (9), which is applied for all working days along the time horizon of 1 year.

$$\sum_k \left[\varepsilon_1(k) + \varepsilon_2(k) \right] \le 0.4\,, \qquad k \text{ – working hours during a working day} \tag{9}$$

Besides Eqs. (8) and (9) also Eqs. (1), (2), (3), (5), and (6) are imposed as optimization constraints, as well as a passive cooling feasibility constraint, for which the description follows.

For periods, during which passive cooling and active cooling are needed to work simultaneously in serial connection, the question arises what is the optimal order in which these two installations should operate in order to supply the required cooling to the TABS. In an initial stage the passive cooling feasibility is defined by Eq. (10), where the temperature difference of 3°C between the supply water and the brine is always guaranteed.

$$0 \le T_{BRINE}(i) \le T_{WS}(i) - 3°C \tag{10}$$

This means that passive cooling can be engaged either as first or second cooling device, however, in both cases it will transfer heat with the same efficiency. On the contrary, active cooling is more efficient when engaged as first device, due to the lower temperature difference, in that case, between the condenser- and evaporator side of the chiller. Thus, by applying Eq. (10) as optimization constraint we assume active cooling as the first installation in the serial connection and passive cooling as the second one (Figure 2).

In practice it is possible to apply passive cooling as the first installation in the serial connection until the temperature difference between the supply water and the brine reaches 3°C and afterwards apply active cooling to further decrease the supply water temperature even below the brine temperature. This configuration obeys Eq. (11) as optimization constraint.

$$0 \le T_{BRINE}(i) \le T_{WS,PC}(i) - 3°C \tag{11}$$

In Eq. (11) $T_{WS,PC}$ represents the supply water, as given in Eq. (12), which is pre-cooled by passive cooling, before applying active cooling.

$$T_{WS,PC}(i) = \frac{-\dot{Q}_{PC}}{UA_{T_{WS}T_{CC}}} + T_{CC}(i) \tag{12}$$

Thus, by applying Eq. (11) (and Eq. (12)) as optimization constraint we assume passive cooling as the first installation in the series and active cooling as the second one (Figure 3).

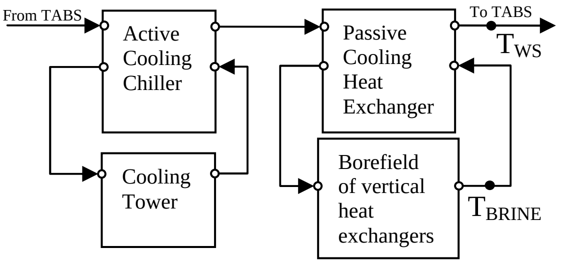

Figure 2. Active Cooling first in the serial connection

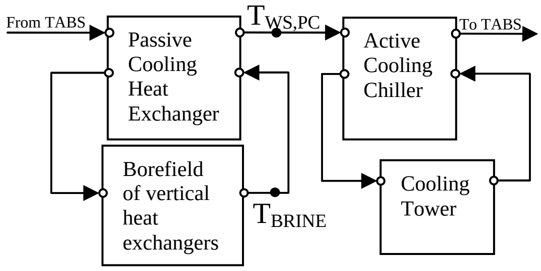

Figure 3. Passive cooling first in the serial connection

In order to investigate long term thermal effects on the ground storage volume, three experiments are carried out, which will be referred to as "Case G1", "Case G2", and "Case G3". In Case G1 the states of the BHE model (ground temperatures $T_j(i)$, $j=1,2,3$) are all initialized to coincide with the undisturbed ground temperature ($T_{GR}=10°C$), as if the borehole has been installed very recently ($T_j(0)=T_{GR}$). The final ground temperature values after 1 year of operation ($T_j(8760)$) are left unconstrained, thus to be optimized. In Case G2 the initial ground temperature values are set to coincide with the final optimal solution (at $i=8760$) from Case G1, as if the borehole has been exploited already for one year. In Case G2 the final ground temperature values are also unconstrained, thus determined by the optimization. In Case G3 annually cyclic boundary conditions are imposed on the ground temperatures ($T_j(0)=T_j(8760)$). The exact annually repeated values are left to be optimized, so that only the cyclic behavior is imposed.

The influence of the Active Cooling capacity on the system's control profiles and annual primary energy use is investigated for the above mentioned downsized borefield (17 BHEs). The AC installation capacity is varied in the range [43, 127 (kW)], for which the system is still able to maintain the required thermal comfort in the office zone by redistributing the loads over all devices.

3. Results and discussion

The results presented in this paper are described in five steps by applying the SALTO method for different system configurations and conditions: (1) Initially sized system (GCHP system with 33 BHEs); (2) Downsized borefield (Hybrid GCHP system with 17 BHEs); (3) Alternative operation of Passive- and Active Cooling installations; (4) Long term balance of the ground storage volume; (5) Downsized Active Cooling capacity.

In the first step SALTO is performed for the initial system configuration, for which the heating and cooling loads could be entirely covered by the borefield, by means of heat pump heating and passive cooling. Active cooling is applied as first cooling installation in the serial connection to the passive cooling one (Eq. (10), Figure 2). Cyclic boundary conditions are imposed on the ground temperatures. The results for the office zone air temperature T_Z and its comfort bounds are depicted in Figure 4. It can be seen that T_Z is kept between the comfort boundaries. Exceptions are on the one hand the periods leading to the daily allowed 0.4 Kh of thermal discomfort during the occupation hours. This leads to a total annual discomfort of 80 Kh while the standard allows 104 Kh for an office Category II. On the other hand there are deviations during periods without occupancy of the office when comfort evaluation is not a criterion.

The optimal profiles for $\dot{Q}_{HP}(i)$ and $\dot{Q}_{PC}(i)$ are depicted in Figure 5. The heat pump has an on/off control profile due to the linear term of operation cost in the objective function. On the contrary, although having a linear operation cost term in the objective function too, the passive cooling operation not always hits its capacity upper bound. Instead, the PC is dictated by the constraint represented by Eq. (10) – the brine temperature rises to a level, above which passive cooling is not feasible.

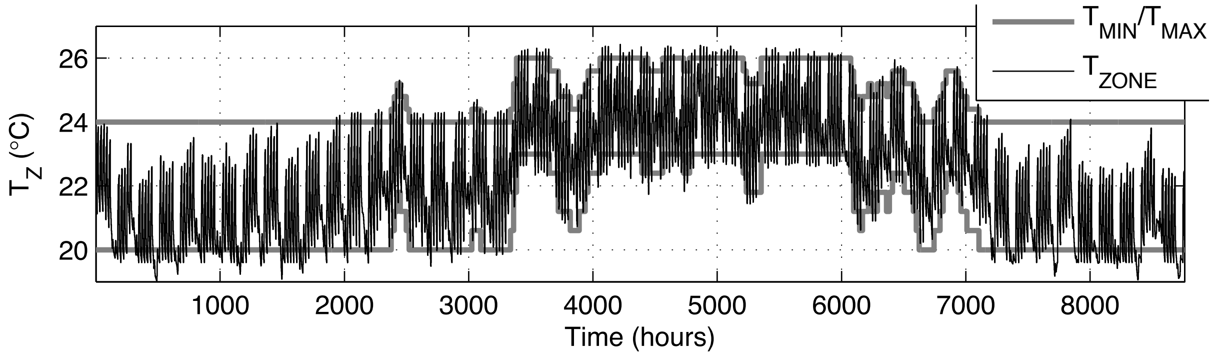

Figure 4. Optimal profile for office zone air temperature

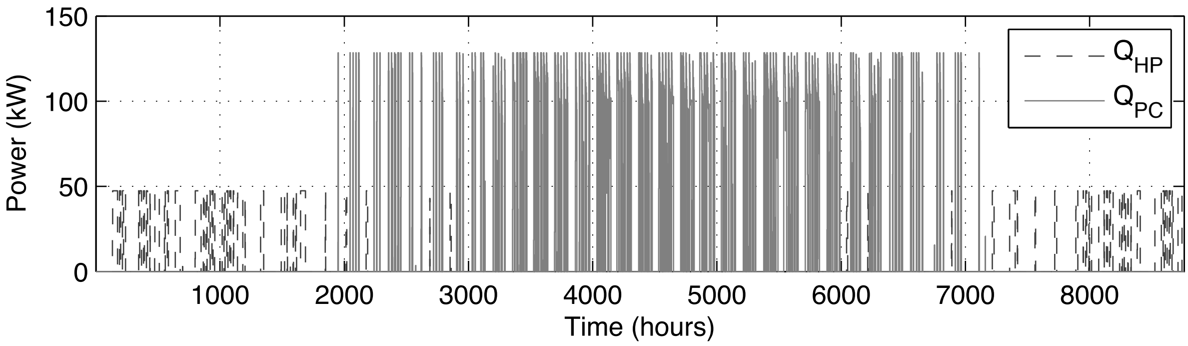

Figure 5. Heat Pump heating and Passive Cooling optimal operation profiles

The temperatures of the ground storage volume, shown in Figure 6, are annually cyclic but their mean values are above the undisturbed ground temperature due to the cooling dominated loads. The annual net heat injection to the borefield amounts to 96.6MWh, which dissipates in the ground adjacent to the storage volume.

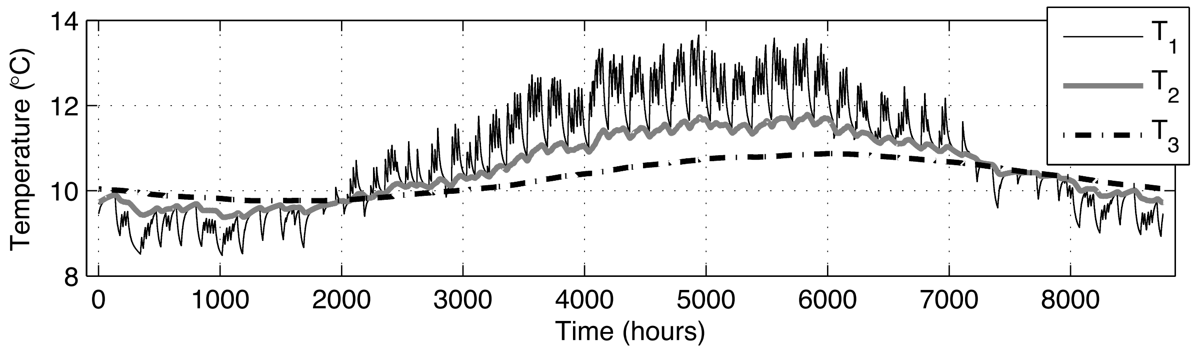

Figure 6. Ground temperature profiles in the ground storage volume

In the second step the borefield is downsized from 33 to 17 BHEs, requiring the back-up devices to compensate. The optimal profiles for ground coupled heat pump operation, passive cooling, and active cooling are depicted in Figure 7. Similarly to the heat pump for heating, the active cooling has also an on/off control profile due to the linear term of operation cost in the objective function. Auxiliary heating is not required due to the cooling dominated office loads.

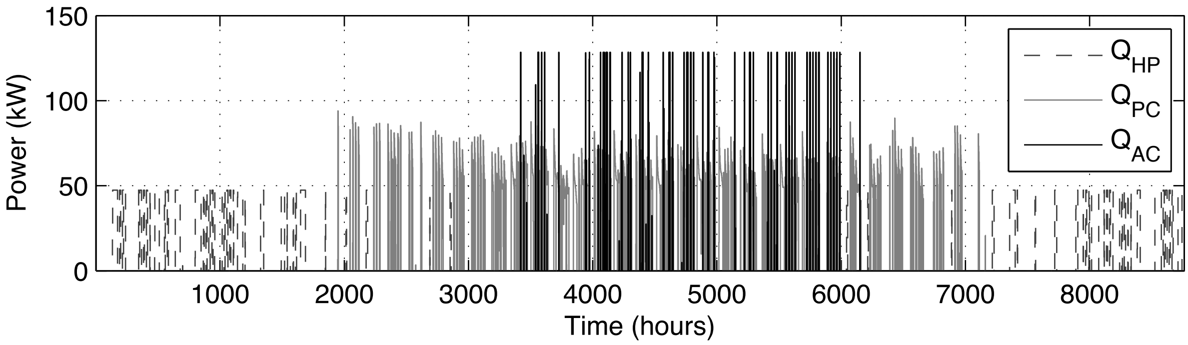

Figure 7. Heat Pump heating, Passive Cooling, and Active Cooling optimal operation profiles

The annual ground temperature profiles are similar to the ones from the previous step (Figure 6) but their magnitude is increased. The annual net heat injection to the borefield amounts to 77.3 MWh.

Due to the fact that active cooling acts before passive cooling, when simultaneous operation of the two installations is needed, a particular profile is observed, which is depicted in Figure 8 and Figure 9.

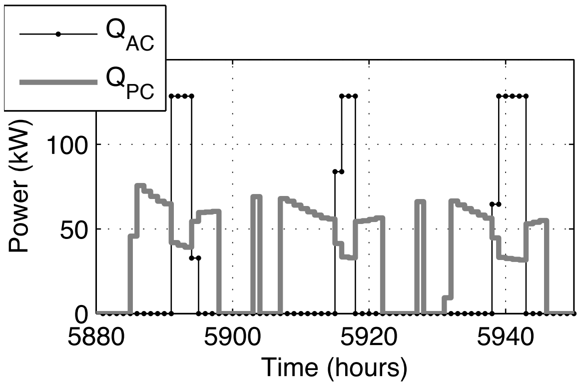

Figure 8. Simultaneous cooling devices operation. Passive cooling as second device

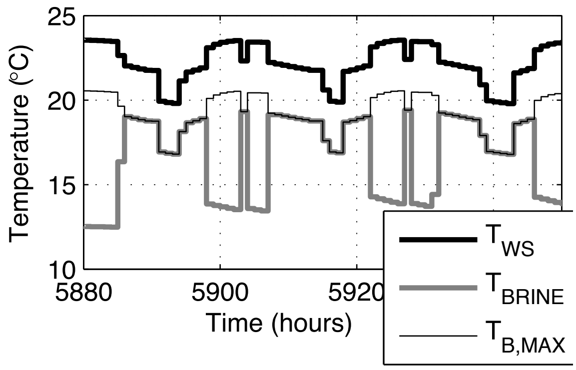

Figure 9. Passive Cooling heat exchanger temperatures. Active Cooling is already applied as first device

In any case, the system should be able to cover the required cooling peaks while keeping the minimal difference between T_{WS} and T_{BRINE} (Eq. (10), Figure 2). In the period of simultaneous operation the passive cooling power decreases (Figure 8). The reason behind is that the water coming from the TABS (Figure 2) is already pre-cooled (by active cooling) when entering the passive cooling heat exchanger. The optimal solution is that the cooling peak is covered by increasing the share of active cooling, while the share of passive cooling is decreased to satisfy Eq. (10). In Figure 9 $T_{B,MAX}$ represents the maximal T_{BRINE} according to the minimal difference relative to T_{WS}.

In the third step the set up from the previous one is inherited, except for the passive cooling feasibility constraint. Passive cooling acts now as the first device and active cooling – as the second one in the serial connection (Eq. (11), Figure 3). Representative fragments of the annual system operation are depicted in Figure 10 and Figure 11. It can be observed (Figure 10) that the passive cooling power does not decrease during the simultaneous operation and the active cooling delivers less power compared to the alternative system configuration (Figure 8). The difference between the supply water temperature and the brine temperature decreases below 3°C (Figure 9) but this is caused by Active Cooling as second device after Passive Cooling as first device, and thus represents no problem. Although active cooling is less efficient when applied as second installation in the series, the annual load it covers is decreased by 11 % in that case, while the share of Passive Cooling, which is much cheaper, is increased. As a result the annual primary energy use for the entire system operation is reduced by 1.3 %.

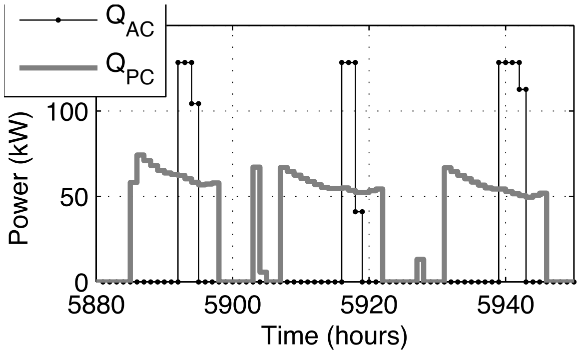

Figure 10. Simultaneous cooling devices operation. Passive cooling as first device

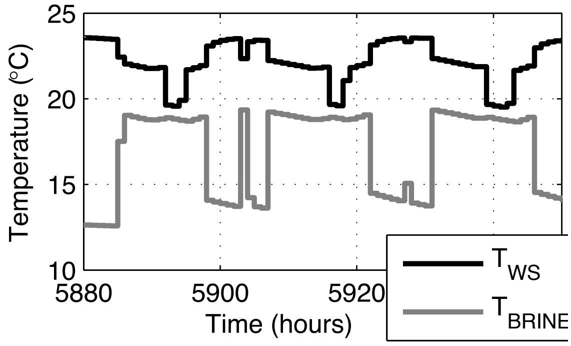

Figure 11. Temperature difference after applying Active Cooling. Passive Cooling is already applied

The fourth step focuses on long term aspects of the ground storage volume exploitation, which are investigated for Case G1, Case G2, and Case G3, described in the end of Section 2. In Case G2 it can be observed in Figure 12 that starting from the final ground temperatures from Case G1 the system operation leads naturally to the same ground temperatures at the end of the year without imposing cyclic behavior. Additionally, the ground temperature profiles from Case G2 coincide with the profiles from Case G3 where the cyclic boundary conditions are explicitly imposed. Furthermore, in Case G1, even starting from the undisturbed temperature the ground temperatures converge to their cyclic profiles during the very first year of operation. Moreover, those three results hold for all system configurations considered in this study.

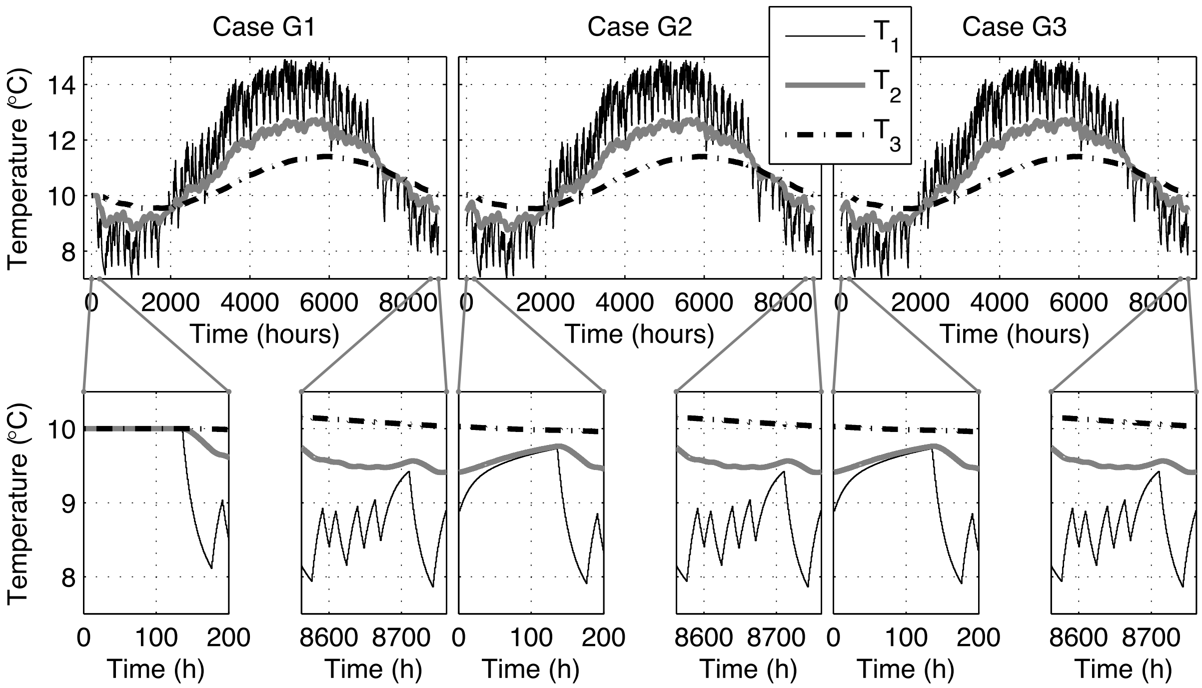

Figure 12. Ground temperatures in the three investigated cases: Case G1 (temperatures initialized at the undisturbed ground temperature); Case G2 (temperatures initialized at the final ones from Case G1); Case G3 (cyclic boundary conditions imposed on the ground temperatures but the exact values are determined by the optimization).
Top: yearly profiles. Bottom: zoom of initial and final periods.

In the fifth step the influence of the active cooling installation capacity on the total annual primary energy use of the system is investigated (Figure 13), for the system with downsized borefield (17 BHEs). The results show that the range of feasible AC capacity limitations can be divided into three regions. AC installation capacities in the range [65, 127 (kW)] correspond to typical active cooling operation, as depicted in Figure 10. The AC operation profile represents a pulse, the width of which increases when the installation capacity decreases. If the pulse is short enough, it is applied simultaneously with passive cooling. Capacities in the range [50, 65 (kW)] require longer pulse widths. AC is started before passive cooling starts in order to provide the required amount of cooling in time. Capacities in the range [43, 50 (kW)] are not large enough to cover the peak cooling period in simultaneous operation with passive cooling. Then, the optimal solution is that before the peak cooling period the ground is pre-cooled by heat pump heating, simultaneously operating with active cooling to avoid overheating of the office zone – Figure 14. Thereafter, the pre-cooled ground unlocks storage for increased passive cooling, which in combination with the limited active cooling is large enough to cover the cooling peak. The heat pump operation added for pre-cooling of the ground causes a significant increase in the annual energy use (Figure 13).

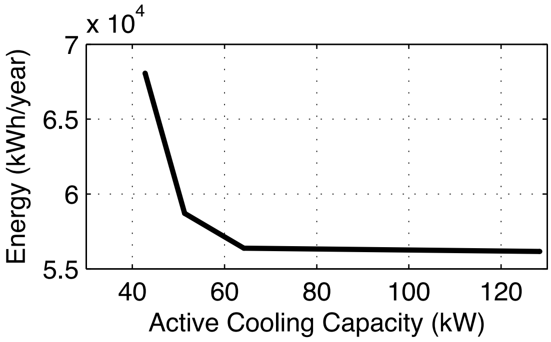

Figure 13. Influence of Active Cooling installation capacity on system total annual primary energy use

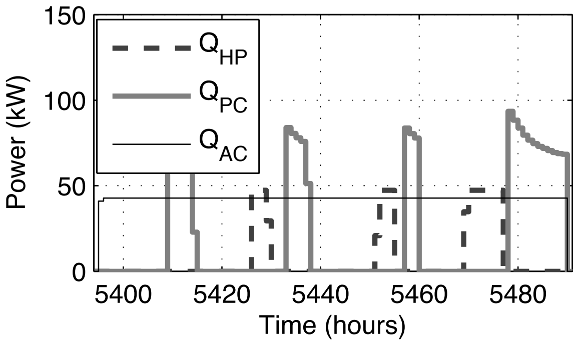

Figure 14. Optimal operation of HP, PC, AC devices, in the case of very small AC capacity range [40, 50 (kW)]

4. Conclusions

The wide range in time scales associated with GCHP systems asks for the combination of short and long term objectives when developing an optimal control strategy. This paper presents an optimization of the heat and cold supply to TABS office zones based on weather and occupancy profiles for a reference year to achieve three main goals: (1) satisfying thermal comfort requirements in the office zones, (2) applying control signals that minimize the system operation cost, (3) retaining long term thermal balance of the ground storage volume. Developing and solving a combined short and long term optimization (SALTO) allows analyses of the distribution of heating and cooling duties between the corresponding devices, guaranteeing a long term thermal balance of the ground storage volume, and thermal comfort inside the building at a minimal energy cost.

The long-term optimal solution, i.e. the control profile which satisfies the imposed thermal comfort level at the lowest annual energy cost and which can be repeated unaltered year after year (for the given weather and occupancy profile), is found by imposing cyclic boundary conditions on the ground storage temperatures. The results indicate that, first, -for a cooling dominated building-, the optimal operation is characterized by an annual net heat injection to the ground and mean ground storage temperatures which are higher than the undisturbed ground temperature. Second, passive cooling is constrained by the upper limit on the brine fluid temperature. Third, the corresponding optimal ground storage temperature distribution is reached during the first year of operation (i.e., starting from ground storage at undisturbed ground temperature). These conclusions hold for all borefield sizes, as well as for all sizes of the supplementary cooling device, which are considered in this paper.

A trade off could be calculated between investment cost for borehole heat exchangers, investment cost for back-up cooling installation, and operation cost of the hybrid system.

Engaging passive cooling as first installation and active cooling as second installation in the serial connection is the cheaper of the two options when both installations are needed to run simultaneously.

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